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Design and implementation of two-degree-of-freedom tuned mass damper in milling vibration mitigation

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ABSTRACT

The tuned mass damper (TMD) has been applied to the machining vibration control widely, and it is categorized into several groups depending on the available degrees of freedom (DOF). Previous works have been mostly focused on the application of single-DOF TMD, but it is revealed that the damping performance could be further promoted by multiple-DOFs TMD. A two-DOF TMD for the milling vibration mitigation is investigated. The TMD possessing translation and rotation motion is designed with tunable stiffness and damping, and the design parameters are optimized numerically based on the H_∞ criterion. The TMD is implemented on a workpiece fixture with single dominant mode, and the experimentally tuned frequency response function (FRF) has 80.8 percent reduction on the amplitude of the flexible mode. Spindle speeds corresponding to the resonance and chatter vibrations are selected for the machining tests. The measured vibrations and surface quality validate the improvement of the machining stability by the TMD, and the critical depth of cut is increased at least two folds.

1. Introduction

Machining vibrations (i.e. resonant vibration, chatter vibration) restrict the cutting efficiency, product quality and tool life. Resonant vibration induced by intermittent cutting force is extremely harmful for machine tool components or workpiece with low damping, and chatter vibration is a major issue in high-speed machining. Chatter is generally classified into primary (i.e. by friction between the tool and workpiece, by thermo-mechanical effects on the chip formation or by mode coupling) [1,2] and secondary (i.e. by the regeneration of wavy surface finish) [1,3]. The occurrence of regenerative chatter has been the most common in the industry, and its avoidance has been investigated a lot. So far, a great deal of strategies have been proposed to mitigate the machining vibrations, including cutting parameter selection based on the stability charts and active or passive control technologies to modify the structural dynamic properties.

Active control is usually composed of monitoring, diagnosis and execution elements, and is well-adapted to the variations of machining process. Rashid developed an active controlled palletized workholding system based on the adaptive control algorithm and piezo-actuators, while the traditional approach is controlling vibrations from the cutting tool or spindle [4]. Munoa developed a biaxial active actuator for the heavy duty milling operations, and demonstrated that the direct velocity feedback law was the most effective among the four analyzed control strategies [5]. Despite the successes in

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attenuating the machining vibration, active control exhibits the disadvantages of high cost and complicated implementation.

Passive control, on the other hand, is effective in the vibration control and easy to implement. The tuned mass damper (TMD) is one kind of widely used passive control devices in the machining. Duncan [6] and Rashid [7] investigated the application of single-degree-of-freedom (SDOF) damper in milling. Moradi studied the optimum values of absorbers' position and springs' stiffness for various milling conditions such that the cutting tool vibration was minimized [8]. Saffury obtained analytically the frequency response function (FRF) of a non-uniform cantilever beam with multiple dampers by the functional perturbation method, and derived the optimized morphology of a general heterogeneous beam with single damper [9]. Sims proposed an analytical tuning criterion for the SDOF TMD in order to achieve the optimum chatter suppression [10]. Wang designed a nonlinear TMD equipped with an additional series friction-spring element, and proved that the critical cutting depth can be largely improved against the linear TMD [11].

Above works are focused on the application of SDOF TMD. However, research works reveal that the damping performances could be further improved by utilizing multiple DOFs of the TMD, i.e, increasing the number of TMDs or designing multi-DOF TMD. Yang presented the optimization routines of multiple-SDOF TMD to increase the chatter resistance of tool clamping element [12]. Yutaka attached three identical dampers to a rotating collet chuck in the end milling operation, and the optimal natural frequency and damping ratio of the dampers for maximizing the chatter-free axial depth of cut were varied with spindle speed [13]. Kolluru utilized six tuned viscoelastic dampers to minimize the vibration of thin wall casings, and the root mean square value of vibration was reduced about 77 percent [14].

Another approach to enhance the damping performance is to utilize more than one DOF of a TMD. Lei Zuo proposed the two-DOF TMD which yields better performance than two SDOF TMDs of equal mass [15]. Jang proposed a method of optimizing the parameters of the two-DOF TMD based on the fixed points, which is easier and more efficiency than numerical optimization [16]. However, design and experimental tuning of the TMD are more difficult as the increase of the DOFs, and application into the machining process of the two-DOF TMD has been quite restricted according to the literature review.

In the present study, the design and implementation of a two-DOF TMD for the milling vibration mitigation is investigated. The TMD with tunable stiffness and damping is designed with translation and rotation motion, and the optimum tuning parameters are acquired based on the numerical optimization approach. The two vibration modes of the TMD are utilized to damp the dominant mode from the workpiece fixture, and the enhancement of machining stability is evidenced. The paper is organized as follows: Equations of motion are formulated and the minimax approach is employed to optimize the TMD based on the H_∞ criterion in Section 2. The structural design of the two-DOF TMD is presented and the experiment FRF tuning is demonstrated in Section 3. Cutting tests corresponding to the resonance and chatter vibrations are carried out to validate the vibration mitigation in Section 4. The paper is concluded in the end.

2. Theoretical formulation of the TMD optimization

2.1. Equations of the motion of the TMD

An illustration of the two-DOF TMD damping the SDOF primary structure is shown in Fig. 1 [15]. The attached two-DOF TMD exhibits two planar degrees of freedom, i.e. translation x_d and rotation θ_d. The primary structure has a mass m_0, stiffness k_0 and viscous damping c_0. The TMD mass is m_d and the radius of gyration is ρ; therefore, the rotational inertia relative to its geometrical center is $I_d = m_d \rho^2$. d is the distance from the center to the spring connections.

Assuming that the primary structure is subjected to an external disturbance force $F_0 e^{j\omega t}$, the equation of motion is expressed as

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\} \tag{1}$$

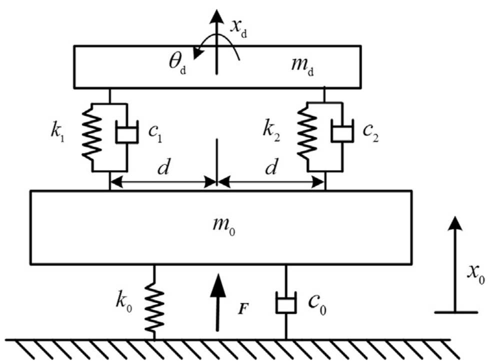

Fig. 1. SDOF primary structure damped by the two-DOF TMD.